HEAT TRANSFER COEFFICIENTS OF STEAM WATER MIXTURES

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AND
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OF

STEAM WATER MIXTURES

by

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PREFACE

With the advent of nuclear power for propulsion of ships and for future industrial applications, an ever increasing importance is being placed on the phenomena of heat transfer toward the attainment of higher thermal efficiencies. It is likely, in connection with nuclear power, that steam will remain the primary means of transmitting nuclear energy into useful mechanical work. Therefore, it is believed that for optimum design of heat transfer equipment as much knowledge as possible about heat transfer to steam-water mixtures is essential.

The accepted index of heat transfer between a solid and a fluid has become known as the surface or film heat transfer coefficient. The authors during their course of instruction at the U. S. Naval Postgraduate School became aware of the limited amount of data available for these coefficients as applicable to steam-water mixtures. An investigation at this same institution carried out in 1953 by Lieutenants Fisher and King (5) produced some pertinent data. The purpose of the present thesis was to extend the range of steam-water mixtures in order to find optimum coefficients for various mass rates of flow.

The experimental work was carried out at the U. S. Naval Postgraduate School from February to April 1954.

It is the authors' desire to express their appreciation to Professor E. E. Drucker for his invaluable aid in the preparation of

the thesis, to the Research Division of the Babcock and Wilcox Company for instrumentation of the test section, and to N. V. DuCette, BTC, USN, and E. A. Shoemaker, MM1, USN, for their aid in construction of the experimental set-up.

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TABLE OF SYMBOLS AND ABBREVIATIONS

A	Area of heat transfer surface, square feet
c_p	Specific heat at constant pressure, Btu/(lb)(deg F)
d	Diameter, feet
G	Mass velocity, lb/(hr)(sq ft of cross section)
h	Coefficient of heat transfer between fluid and surface, Btu/(hr)(sq ft)(deg F)
k	Coefficient of thermal conductivity, Btu/(hr)(ft)(deg F)
kw	Kilowatt
L	Heated length of tube, feet
M	Mass flow rate, lb/hr
u	Fluid Viscosity, lb/(hr)(ft)
P	Total pressure, lb/sq in
π	3.1416
ρ	Fluid density, lb/cu ft
q	Rate of heat transfer, Btu/hr
r	Radius, feet
t	Temperature, degrees Fahrenheit
Δt	Temperature difference, (deg F)
v	Fluid Velocity, ft/hr
Nu	$\frac{hd}{k}$, Nusselt Group, dimensionless
Pr	$\frac{uc_p}{k}$, Prandtl Group, dimensionless
Re	$\frac{\rho vd}{u}$, Reynolds Group, dimensionless

Subscripts:

d depth i inner o outer s surface

INTRODUCTION

The objective of this thesis was to investigate the variation of the heat transfer coefficient of steam-water mixtures in the range of 10 to 50 percent moisture and also to determine the quality at which this index of heat transfer has a maximum value. A secondary objective was to verify the variation of h with mass rate as reported by Fisher and King as well as to extend the investigation to much higher mass rates.

The test section used by Fisher and King was employed in this investigation. The desired qualities were obtained by injecting water in a fine spray into the steam line. This mixture was then passed through the test section. Runs were made under the following conditions: pressures from 40 psia to 150 psia at the test section inlet and moisture content from 0 to 45 percent.

The piping from the outlet of the test section to the condenser was designed to provide for the necessary pressure drop without having choking occur anywhere in the line. By this means the maximum mass rate was increased from 230 pounds per hour as reported by Fisher and King to 1100 pounds per hour at the 150 psia inlet pressure condition.

Calibration runs were made at each pressure to determine thermocouple corrections. There runs indicated that some thermocouples were in error by as much as 10° F.

Surface temperatures, extrapolated from corrected depth thermocouple readings, were found to be lower than the steam temperature

at the corresponding cross sections of the test section in most runs. This would tend to lead to the erroneous conclusion that negative values of the heat transfer coefficient existed.

CHAPTER I

TWO PHASE FLOW

Heat transfer in two-phase flow inside tubes has been investigated and data collected, both for vaporization phenomena and for the simultaneous preheating of a liquid and a permanent gas. However, a survey of applicable literature reveals that very little information is available for two-phase flow of gas-liquid mixtures and, in particular, steam-water mixtures.

McAdams (6) and others (7) have found from data for vaporization inside tubes that the local heat transfer coefficients in the preheating section are higher than would be predicted from the widely accepted equation of forced convection through a tube for single-phase flow:

$$Nu = 0.024\ Re^{0.8}\ Pr^{0.4}$$

This phenomenon is believed to be the result of the formation of vapor bubbles prior to boiling. These bubbles disrupt the boundary film locally, and as they move from the wall into the liquid they are condensed, giving up their heat of condensation. This hypothesis would explain the higher coefficients found in the preheating section.

On the basis of this theory it is logical to predict that the film coefficient would increase as a gas is passed through a liquid. This has been shown to be true experimentally, for example, by Verschoor and Stemerding (9) in an investigation of air and water passing upward through a vertical tube. In their investigation a maximum film coefficient was reached when the ratio of air to water was such as to change

the type of flow from slug flow to annular flow. It had been observed by Bergelin (2) in 1949 that if a gas and a liquid pass vertically upward simultaneously in a tube, three distinct flow conditions occur as the gas-liquid ratio is increased, as follows:

(a) Bubble flow, when gas bubbles pass individually through the tube;

(b) Slug flow, when alternate slugs of gas and liquid pass through the tube; and

(c) Annular flow, when a liquid flows in an annulus along the tube wall while gas passes at a much higher velocity through the center of the tube.

Verschoor and Stemerding also found in their air-water investigation that not only does the film coefficient reach a maximum at the change over from slug to annular flow, but also the coefficient may decrease when the flow changes from bubble to slug flow, if the liquid flow rate is low. Their results also show that the lower flow densities produced lower value of heat transfer coefficients.

In 1953 Fisher and King in an investigation of steam-water mixtures presented results which show a marked increase in the local heat transfer coefficient as the moisture content increases. No maximum values of the coefficients were reported in that they were limited to a maximum moisture content of 6%.

CHAPTER II

EQUIPMENT

The equipment consisted of a flow system in which steam was taken from a main line and passed through a throttle valve, 30 feet of piping, the test section, a centrifix water separator, 60 feet of piping, a condenser, and to a weight tank. Water drains from the centrifix unit passed through a small leveling tank, a needle valve, a cooler and to a weigh tank. The schematic flow diagram is shown in Figure 1. Steam supply came from a Babcock and Wilcox FM boiler at a maximum pressure of 200 psig. After passing through the throttling valve the steam passed vertically downward 15 feet through a 2 inch pipe and then vertically upward to the test section as shown. The purpose of this loop was to give the steam-water mixture a long straight path prior to entering the test section in order to achieve thorough mixing.

At the exit of the test section the flow was directed into a centrifix water separator which was mounted vertically to accept downward flow. Primary separation was accomplished in this unit by conical baffles which caused the water to be thrown to the wall of the unit and to collect at the bottom. The steam passed through the center into the exit steam pipe. Water level had to be maintained below the top of this pipe to prevent "spill over" of water into the dry steam line. This was accomplished by installing a small level tank and gage glass in parallel with the centrifix unit and was controlled by a needle valve on the centrifix drain line. The drains from the centrifix

passed through a cooler and into the weight tank.

The moisture content of the steam was determined by a throttling calorimeter immediately following the centrifix. It was found that with a sufficiently large centrifix the moisture content of the steam was below 1% at all times, despite inlet moisture content as high as 45%.

Steam leaving the centrifix passed through 16 feet of 3/4 inch pipe, 35 feet of 1 1/2 inch pipe, 7 feet of 2 inch pipe, and a throttle valve into a condenser operated at atmospheric pressure. This piping was installed in order to dissipate the pressure while maintaining a maximum pressure at the test section outlet. Operation was very satisfactory and as a consequence, mass rates up to 1100 pounds per hour were obtained.

Water was injected by use of a centrifugal vane type pump which took suction from a tank of water, steam heated to 200° F. The water was pumped through a 2 inch section of pipe which contained a 7 kw, 230 volt Chromolox heater, which was controlled by a resistor bank in series with the heater. This heater was capable of heating low flow rates of water to the same temperature as the incoming line steam. The water passed from this heater through a small spray nozzle into the 2 inch steam line following the inlet throttle valve. The flow was controlled by a needle valve in the recirculating line.

The test section was the same as that used by Fisher and King, details of which are shown in figure 3. In general, it consisted of a 24 inch vertical steel tube having an inside diameter of 0.50 inches and an outside diameter of 1.25 inches. Four pairs of chromel-alumel

thermocouples were installed by the Research Division of the Babcock and Wilcox Company. One of each pair was located at the outer surface of the tube and the other at a known depth below the surface. The four pairs were equally spaced along the axis of the section. At either end of the test section iron-constantan thermocouple probes were projected into the fluid stream for measuring the steam temperature.

The heating unit consisted of four independent heating elements containing 30 feet of No. 18 Nichrome V wire insulated with glass sleeving and wound around the tube in a single layer. Each heater was approximately 6 inches long. A thin layer of mica was wrapped around the tube prior to winding the heaters. Power supply to the heaters was controlled by four 115 V - 15 amp powerstats and was measured by portable wattmeters.

Heat insulation of the test section consisted of wrapping the tube outside the heaters with asbestos tape which in turn was covered by magnesia brick. Heat loss was considered negligible through this insulation.

CHAPTER III

OPERATING PROCEDURE

The variables associated with the experimental set-up were heat flux, mass rate of flow, pressure, and percent moisture. The test section heater was limited to 1 kw per heater or a total power input of 4 kw.

The operating procedure began by establishing a mass rate which could readily be estimated from the weigh tanks. The steam-water mixture was fixed by adjusting the amount of water injected, while simultaneously reducing the steam mass rate by throttling either the inlet throttle valve or the condenser throttle valve or both. Water injection was controlled by a needle valve located on the water pump by-pass line. Power was normally set for 1 kw per heater, and pressure at 90 - 120 psig inlet to test section. Steady state conditions were considered to exist when readings of the thermocouples were reproducible for a period of several minutes. Steady state conditions usually existed within 30 minutes provided the system had been initially warmed. The initial warm-up required approximately one hour. In general, thermocouple readings were very constant and all runs are believed to have been recorded under steady state conditions.

Recorded data for each run included thermocouple readings, throttling calorimeter data, pressure at inlet and exit of test section, mass rate data and power input.

CHAPTER IV

METHOD OF CALCULATIONS

The heat transfer coefficient h is defined as the proportionality factor in Newton's law of cooling:

$$dq = h\, dA\, (t_s - t)$$

where dq is the local rate of heat transfer through a surface element dA.

In conducting this investigation, the heat transferred was assumed to be the same as the energy supplied to the test section by means of the heater coils. The rate of energy supplied was measured by means of wattmeters. The area was easily determined by measurement. The temperature of the steam, t, was determined by measuring the pressure at the inlet and outlet of the test section. The pressures at the cross sections containing the depth thermocouples were calculated, based on the assumption that the total pressure drop through the test section was linear. The steam temperature at each position was then taken as the saturation temperature corresponding to this pressure.

The surface temperature, t_s, was determined by extrapolating the temperatures indicated by depth thermocouples 2, 4, 6, and 8 (see figure 3). This was done by computing the temperature drop due to conduction corresponding to the heat input. The temperature drop was calculated using the Fourier conduction equation for steady state heat transmission:

$$q = - k\, A\, \frac{dt}{dx}$$

For a circular tube, this equation may be integrated to give:

$$q = 2\pi K L \frac{\Delta t}{\ln \frac{r_o}{r_i}}$$

The depth thermocouples were used exclusively in the determination of the inside surface temperature because the outside surface thermocouples were subject to possible radiant heat from the power coils. For a power input of 1 kw per coil, the temperature drop due to conduction was calculated to be 10.6° F.

The thermocouples were calibrated by passing steam through the test section with no power supplied to the power coils and allowing the system to reach equilibrium. Since the heat loss was negligible, the thermocouples should have indicated a temperature equal to that of the steam. The difference between these two temperatures varied from 0 to 10° F depending upon the thermocouple and the temperature indicated. A suitable correction was made in all runs before extrapolating for the inside surface temperature.

After passing through the test section, the steam-water mixture was passed through a centrifix to remove the major portion of water in the mixture. The mass rate of the water thus removed was determined by weighing. The remaining steam-water mixture was piped into the condenser and then weighed. A sample of the latter was passed through the throttling calorimeter immediately after leaving the centrifix in order to determine the quality of the mixture condensed. With this data, the quality or moisture content and enthalpy at the exit of the

test section, the total mass rate, and the mass velocity were calculated. The quality of the steam at each section of the test section corresponding to the location of the thermocouples were computed by adjusting the enthalpy of the outgoing steam in an amount equal to the heat input between that section and the exit.

CHAPTER V

CONCLUSIONS AND RESULTS

A number of calibration runs were made to determine thermocouple errors. These runs were made by passing steam through the test section with no power input. After steady state conditions were reached, the temperature at each thermocouple was assumed to be the saturation temperature of the steam at the pressure existing at the corresponding section of the test section. The errors thus determined were reproducible within 2 degrees for any given temperature. The corresponding corrections were applied to all thermocouple readings before calculations were made.

Initial runs were made passing line steam having a moisture content of less than 4 percent through the test section. The values for h obtained from these runs were of the same order of magnitude as those predicted for dry steam as reported by McAdams (6).

$$\frac{hd}{k} = 0.021 \left(\frac{dG}{u}\right)^{0.8}$$

The values for the heat transfer coefficient varied from 400 to 2000 Btu/hr/^{o}F/ft^2 which followed the trend indicated by Fisher and King. These values are included in figures 5 through 8.

The heat transfer coefficients as determined for cold water passing through the test section were within 10% of those predicted by use of McAdams (6) figure 85.

Runs in which water was injected into the steam line indicated

the existence of a negative value for the heat transfer coefficient, which was evidenced by the fact that the extrapolated inside surface temperature was lower than the steam temperature. This phenomenon occurred at sections B, C, and D. (Sections A, B, C, and D being the four sections of the test section from inlet to outlet respectively.) Several factors were considered as possible causes of this negative temperature gradient. The possibility that equilibrium of the steam water mixture had not been reached prior to entry into the test section was considered. Separation of water at the top of the test section with subsequent down flow of water along the inside surface of the test section could account for this error. The fact that the expected temperature difference across the surface film was small, within the accuracy of the thermocouples, was another factor considered.

In an attempt to eliminate the possibility of not having reached equilibrium conditions, the following steps were taken. A spray nozzle which produced a very fine spray was installed at the injection point, oriented to inject the water axially into the steam line parallel to the steam flow. The water was injected approximately 30 feet ahead of the test section. The water was heated before entering the steam line. The injection temperature was varied from 70° to several degrees above the temperature of the steam itself. The results of these runs continued to indicate the existence of a negative heat transfer coefficient. Several runs were then made in which water was injected into the steam while the power was secured. The error of the thermocouples was found to be the same as that found in the previous calibration

runs indicating that equilibrium had been reached and ruling out the possibility of cold water flowing down the sides of the test section.

The fact that section A indicated a positive temperature drop at all times is believed to be a result of the formation of scale deposits at the first heated section. This belief is borne out by the fact that the temperature difference indicated at section A was considerably different from that at sections B, C, and D, while the latter three were much closer together. All positive values of the heat transfer coefficient are shown in figures 5 through 8.

Runs were made at various pressures, mass rates, qualities, injection temperatures and power inputs with similar results. It is believed that the major factor causing these negative values is the limited accuracy obtainable with the present instrumentation. It is felt that further study of this subject is warranted. A possible improvement in design would be the manufacture of a new test section made of copper to reduce the temperature drop through the section. With a maximum power input of 1 kw per coil, the heat flux density was limited to 52,200 Btu/hr/ft^2. By increasing the maximum power input to 5 kw and reducing the inside diameter of the test section to 0.25 inches, the heat flux density would be increased to 522,000 Btu/hr/ft^2, thereby increasing the temperature drop across the surface film by a factor of ten. By calibrating the thermocouples over the entire expected operating range, more precise temperature data could be obtained. Another suggested improvement in the instrumentation would

be the installation of a differential manometer across the test section.

The design of a sight section to enable visual observation of the nature of flow immediately before or after the test section would allow a correlation between the flow condition and changes in the variation of the heat transfer coefficient.

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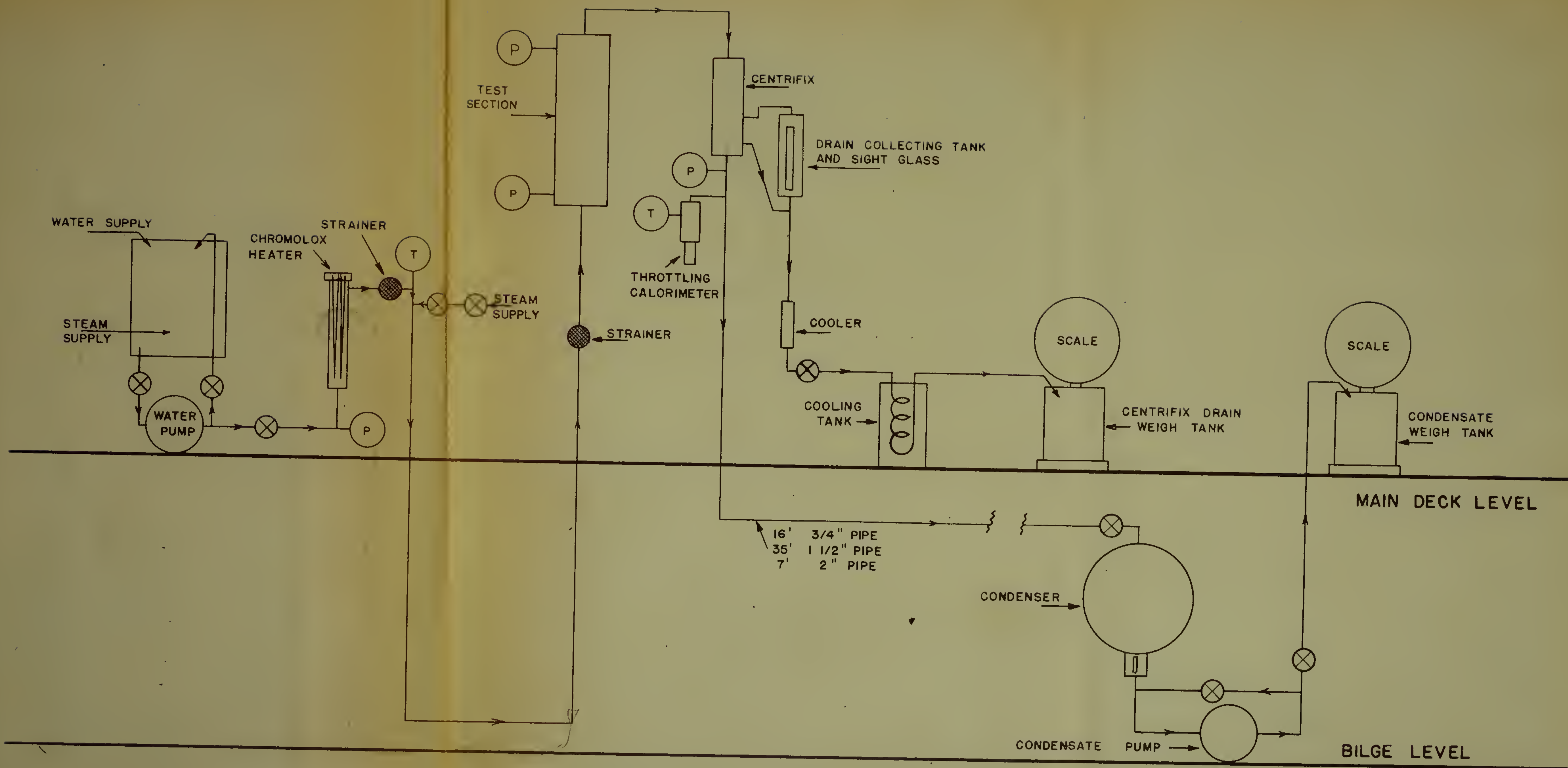
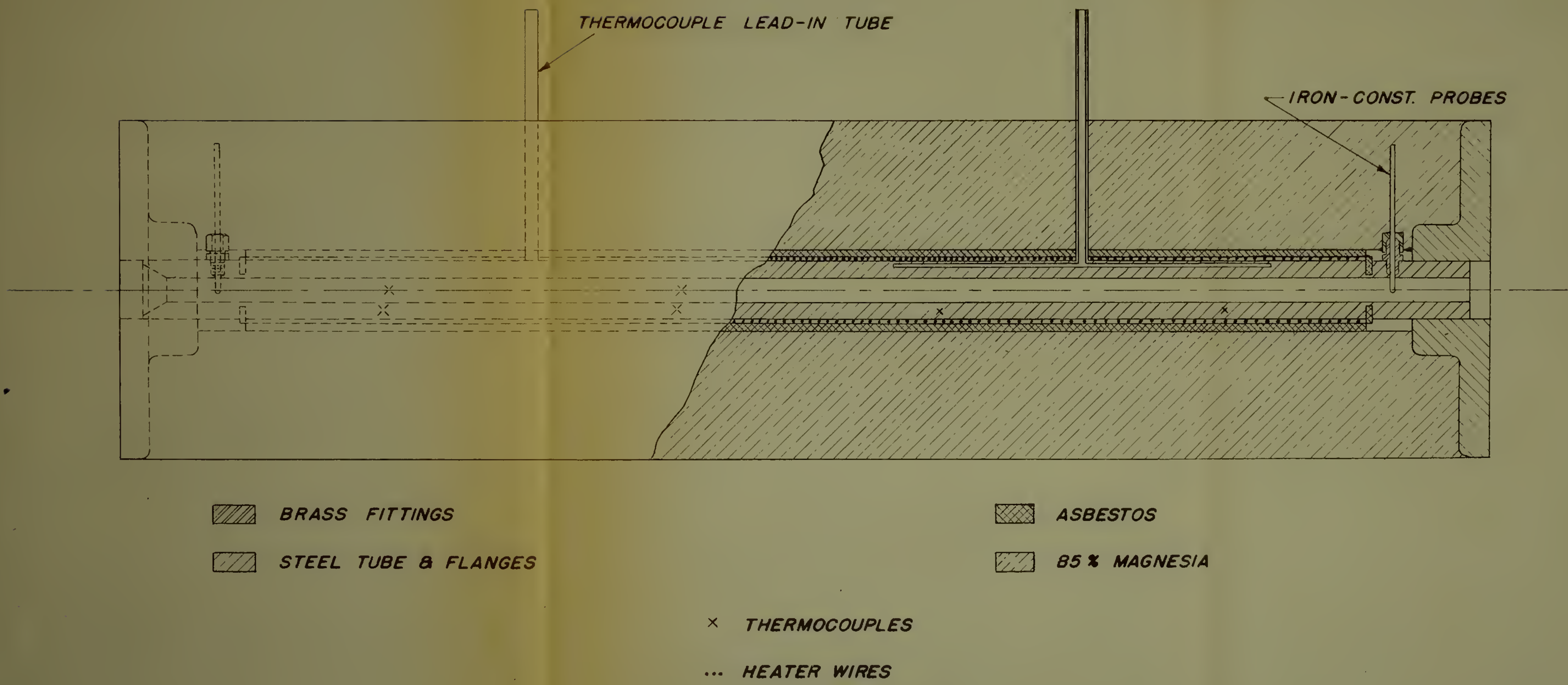


SCHEMATIC FLOW DIAGRAM – FIGURE 1

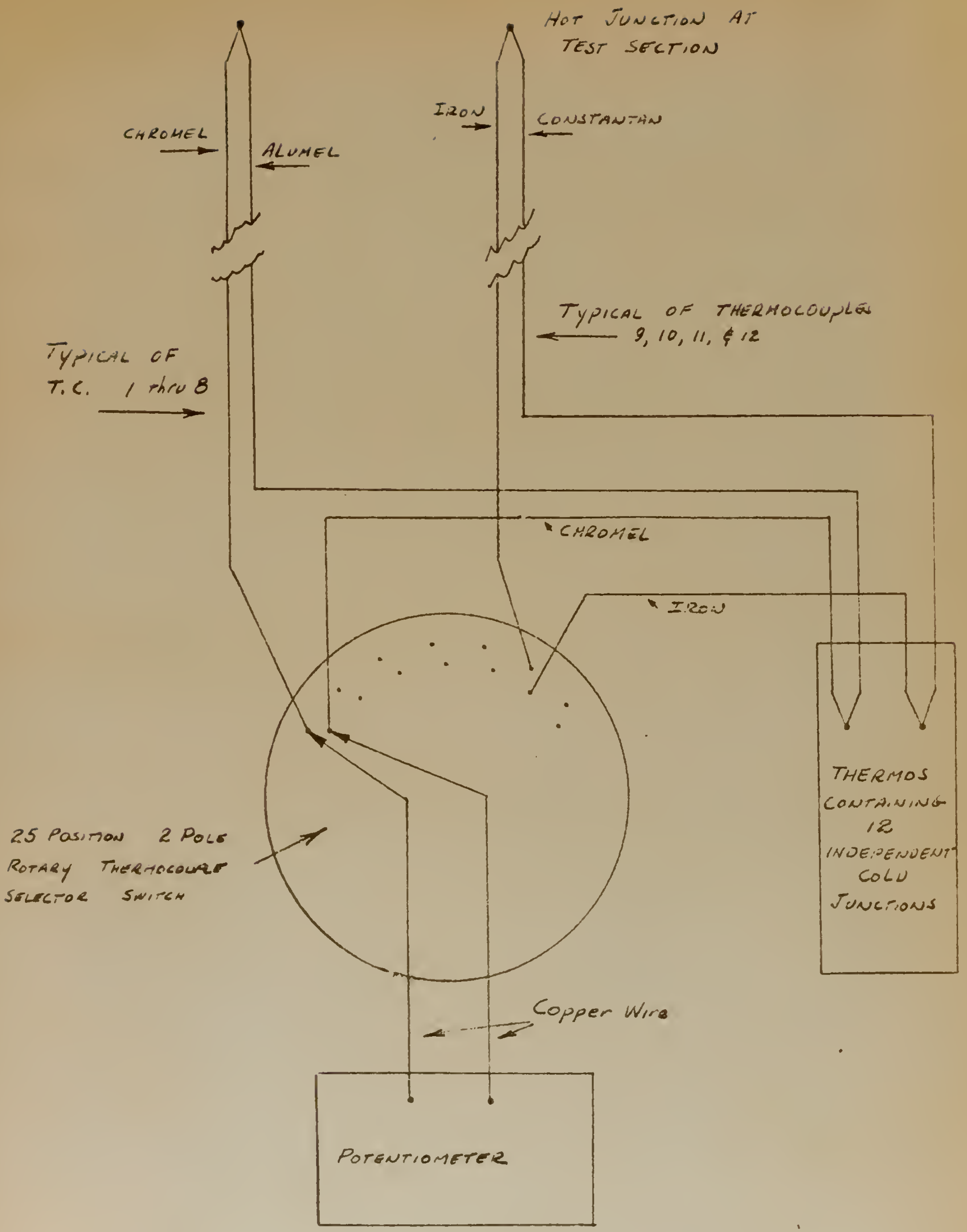


ASSEMBLED TEST SECTION - FIGURE 2

θ	C	H
60.5°	0.6298"	0.0851"
90°	0.8838"	0.1831"
118°	1.071"	0.3031"

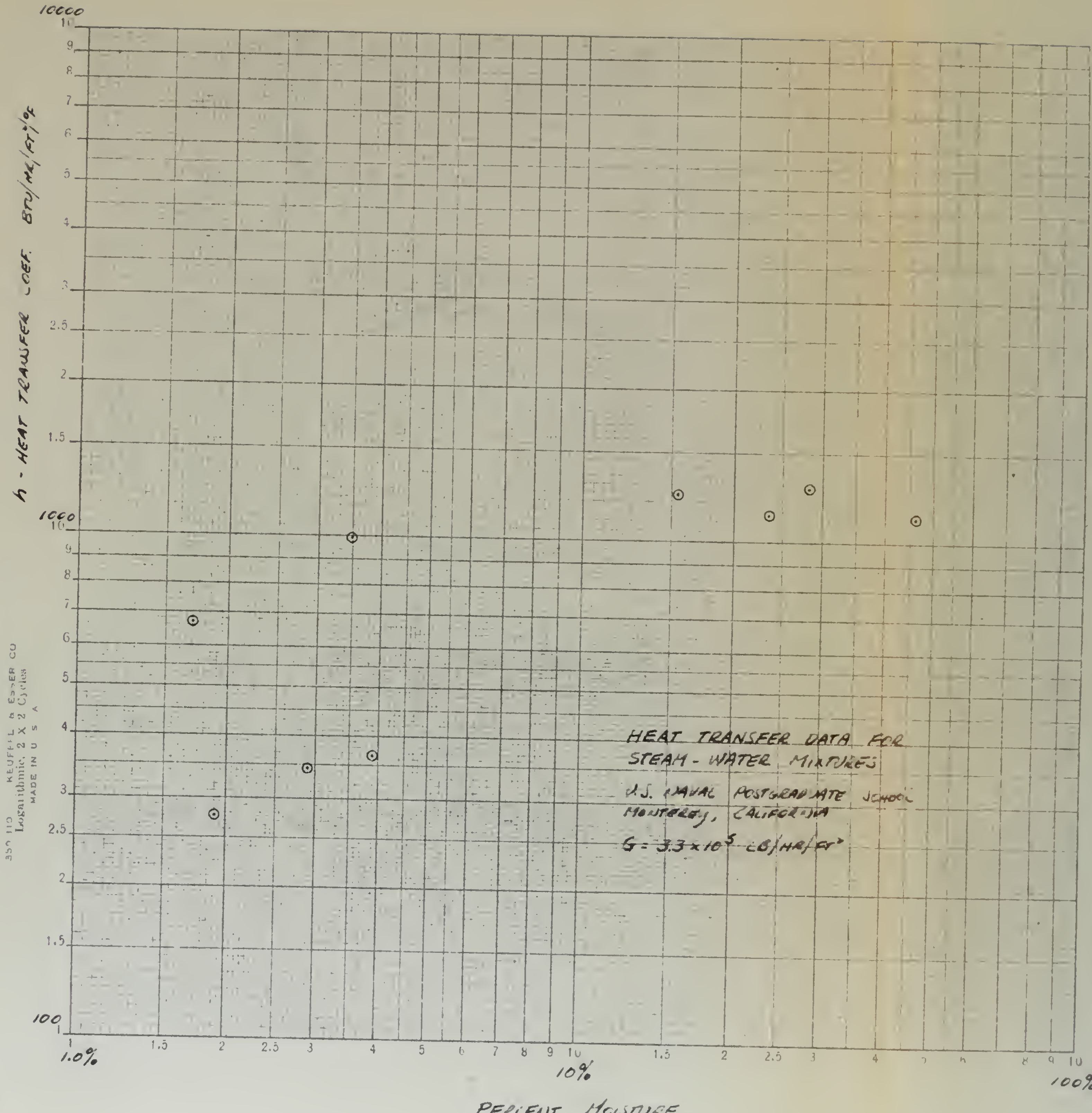
CHORD TABLE

DETAIL OF TEST SECTION — FIGURE 3



SCHEMATIC THERMOCOUPLE CIRCUIT

FIGURE 4



10000
h - HEAT TRANSFER COEF. BTU/HR/FT²/°F
1000
100
1.0%
10%
100%
PERCENT MOISTURE
HEAT TRANSFER DATA FOR STEAM - WATER MIXTURES
U.S. NAVAL POSTGRADUATE SCHOOL MONTEREY, CALIFORNIA
G = 3.3 x 10⁵ LB/HR/FT²
KEUFFEL & ESSER CO
Logarithmic, 2 X 2 Cycles
MADE IN U S A

FIGURE 5

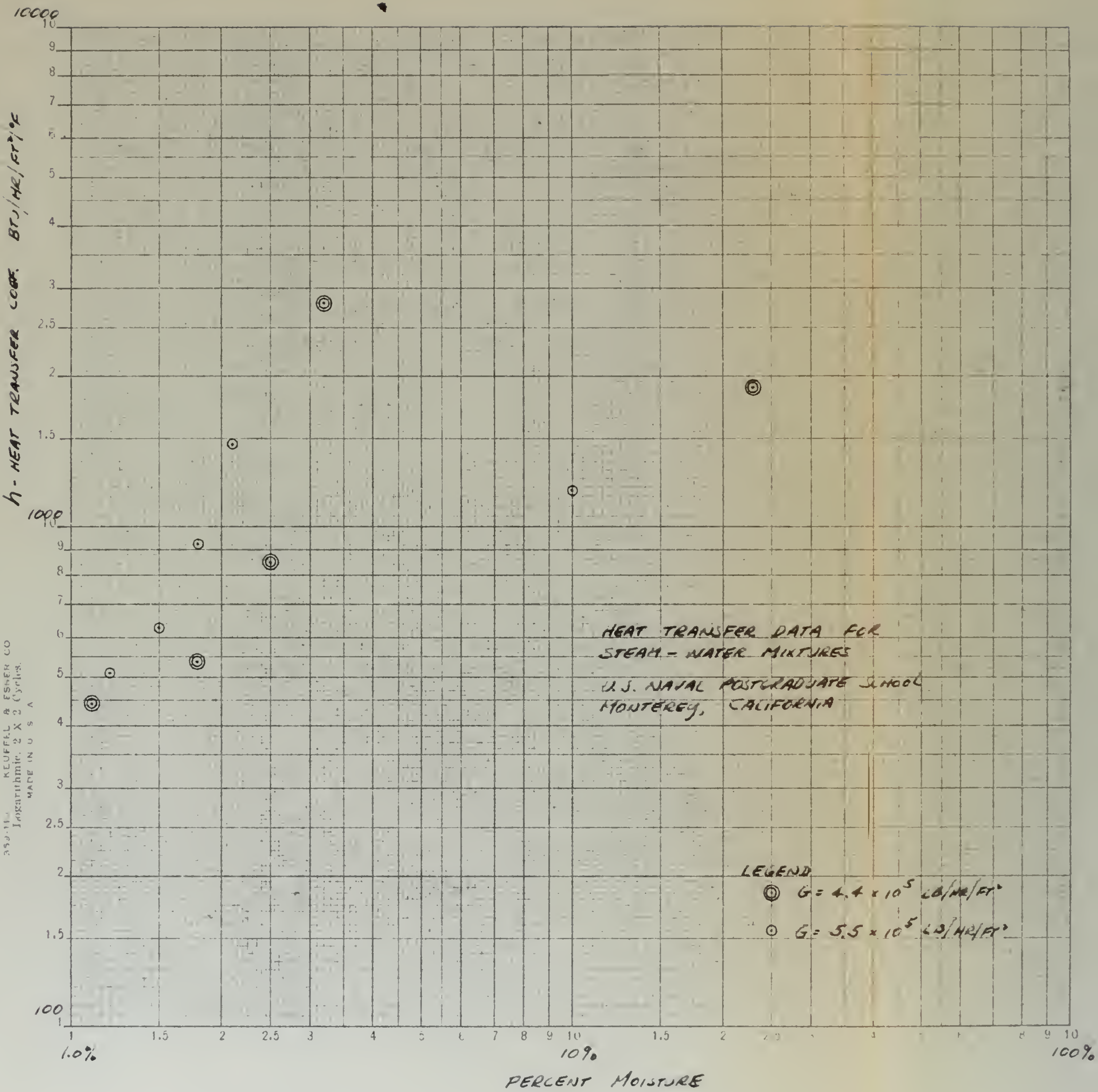

10000
1000
100
h - HEAT TRANSFER COEF. BTU/HR/FT²/°F
1.0%
10%
100%
PERCENT MOISTURE
HEAT TRANSFER DATA FOR
STEAM - WATER MIXTURES
U.S. NAVAL POSTGRADUATE SCHOOL
MONTEREY, CALIFORNIA
LEGEND
G = 4.4 x 10^5 LB/HR/FT²
G = 5.5 x 10^5 LB/HR/FT²
KEUFFEL & ESSER CO.
Logarithmic, 2 X 2 Cycles
MADE IN U.S.A.

FIGURE 6

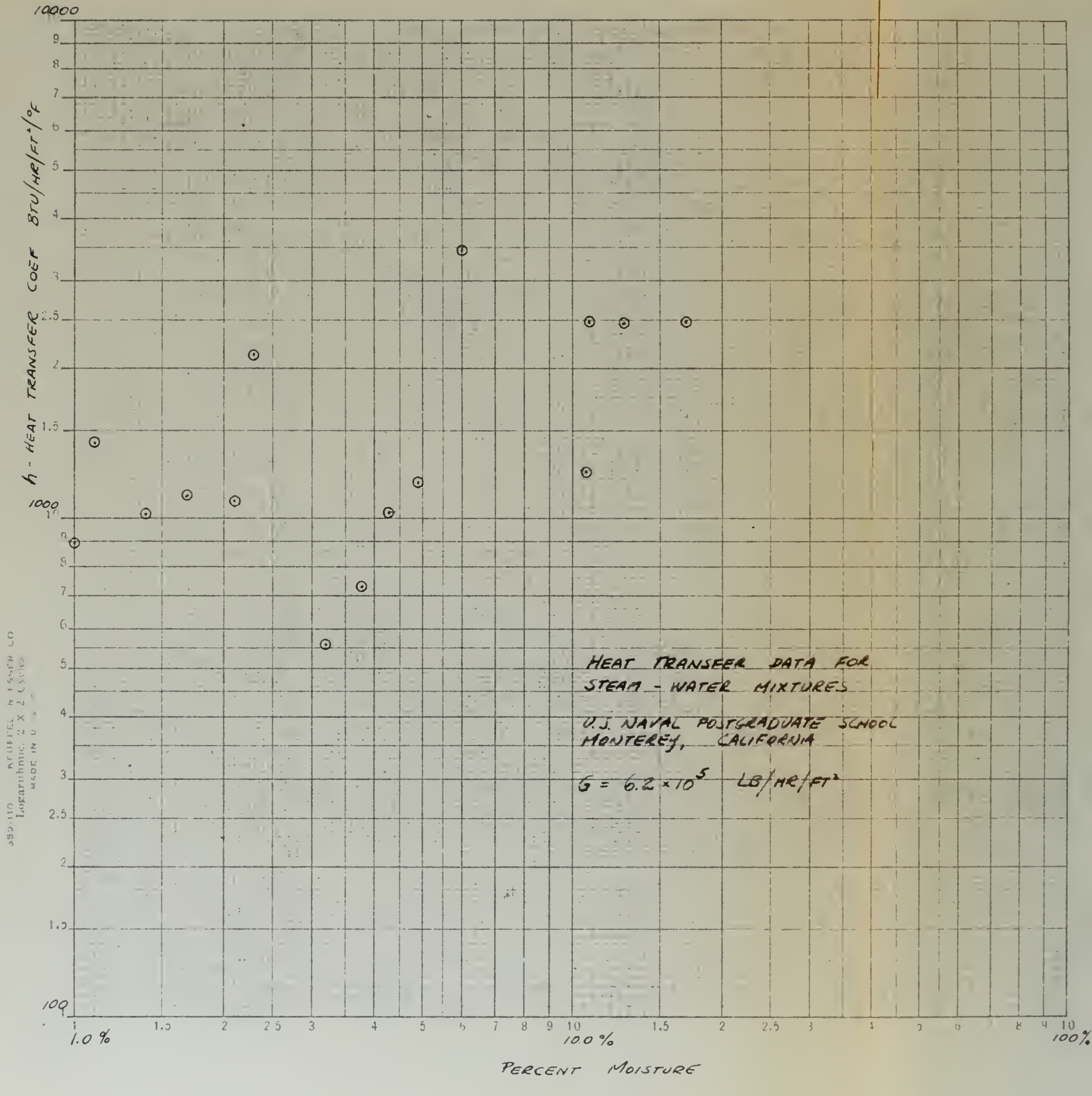

10000
1000
100
h - HEAT TRANSFER COEF BTU/HR/FT²/°F
1.0 %
10.0 %
100%
PERCENT MOISTURE
HEAT TRANSFER DATA FOR
STEAM - WATER MIXTURES
U.S. NAVAL POSTGRADUATE SCHOOL
MONTEREY, CALIFORNIA
G = 6.2 × 10^5 LB/HR/FT²

FIGURE 7

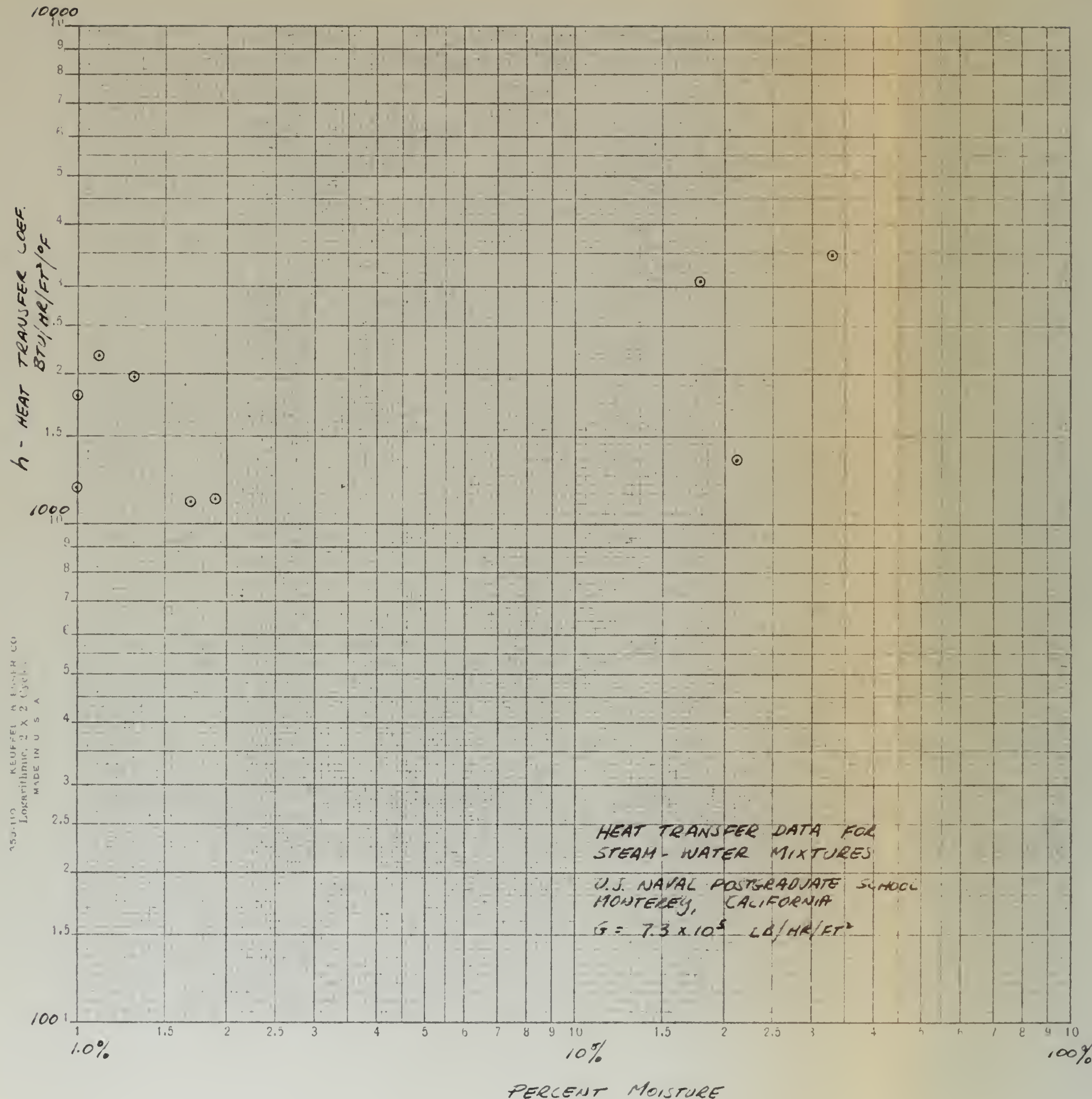

10000
1000
100
h - HEAT TRANSFER COEF.
BTU/HR/FT²/°F
1.0%
10%
100%
PERCENT MOISTURE
HEAT TRANSFER DATA FOR
STEAM - WATER MIXTURES
U.S. NAVAL POSTGRADUATE SCHOOL
MONTEREY, CALIFORNIA
G = 7.3 x 10⁵ LB/HR/FT²

FIGURE 8

25273
Thesis
D1695
Davis
Heat transfer coefficients of steam water mixtures.